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Research Paper

Identifying techno-economic improvements for a steam-generating heat pump with exergy-based cost minimization☆

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ABSTRACT

Steam-generating heat pumps show great potential for reducing carbon emissions in the industrial sector. However, predicting their performance is challenging, as the irreversibilities of components evolve differently with temperature lift and condenser temperature. With over seventy design improvements mentioned in the literature, selecting the most effective design improvement is cumbersome. In this study, energy and exergy-based methods were compared in their ability to identify favourable design changes to a single-stage subcritical heat pump for the generation of steam from hot condensate. The introduction of a sequential compressor with an intermediate cooler, based on the results of the energy analysis reduced the heat pump's techno-economic performance. The results of exergy-based methods lead to the addition of either an internal heat exchanger or a flash vessel by and improved in both cases technoeconomic performance. The internal heat exchanger performed best and increased the coefficient of performance from 2.3 to 2.8 and reduced operational costs by 0.8 M€ after 5 years of operation. Additionally, the initial investment decreased by 135 k€, and the total costs of operation decreased from 10.3 M€ to 8.7 M€. These findings show that exergy-based methods are the way forward in identifying effective design improvements for steam generating heat pumps.

1. Introduction

1.1. Steam-generating heat pumps

Heat pumps are increasingly recognized as a crucial technology for the energy transition [1]. Marina et al. [2] estimated that heat pumps can reduce industrial CO_2 emissions by 30 % when using renewable electricity to upgrade waste streams. However, their deployment is hindered by unfavourable economics compared to other (fossil) heating alternatives [3]. Yet, Marina et al. [2] also identified that most industrial heat is supplied in the form of steam. Utilizing existing steam infrastructure could significantly reduce integration cost, thereby improving the economic feasibility of heat pumps in industry [4].

Various steam-generating heat pumps (SGHPs) are being developed, or have already come to market, leveraging this economic advantage [5,6]. For instance, Marina et al. [7] have developed an experimental SGHP that produces steam at 150°C. Their design consists of a cascade cycle with an intermediate temperature evaporator and internal heat exchangers to superheat suction gas before the compressor of each stage. Similarly, the Kobelco company [8] offers a SGHP that supplies steam at 120°C. This heat pump uses a two-stage compressor with intermediate cooling and an internal heat exchanger to produce hot water. This hot water is thereafter flashed to produce saturated steam. Higher steam temperatures are achieved by successive steam (re)compression [8]. Both the design by Marina et al. [7] and Kobelco [8] are a more advanced version of the common subcritical single-stage (SS) heat pump cycle aimed at improving the techno-economic performance.

1.2. Economics and exergy analysis of steam-generating heat pumps

The SS cycle becomes uneconomical at higher temperature lifts because of increased operational costs from various irreversibilities. Advanced heat pump configurations aim to minimize these irreversibilities which results in less work being required by the compressor and therefore a higher coefficient of performance (COP). The proposed process alterations in advanced cycles involve adding components such as expanders, compressors, flash vessels, mixers, ejectors, heat

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Nomenclature

2CI
2FV
A
C
COP
Ex
f
h
IHX
K
m
q
S
SS
T
t
TCI
TCO
U
W
X

Greek symbols

η
Δ

Subscripts & superscripts

0,p
c
cd
CEPI
d
des
el
evap
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exchangers, or entire top or bottom cycles [9]. Additionally, changing the working fluid could also be used to improve performance [10].

Exergy analysis is a technique that helps to identify where irreversibilities occur within a process. Exergy is defined as the theoretical maximum useful work obtained when a system is brought into thermodynamic equilibrium with the environment by means of processes in which the system interacts only with its environment [11]. Exergy analysis has helped to design heat pumps since its initial conception in the 1960s [11]. However, only a few studies exist for high-temperature heat pumps, while Bergamini et al. [12] showed that the distribution of exergy destruction over the components of the high temperature heat pump cycle differs from low temperature heat pumps. Cao et al. [13] studied six single- and two-stage compression heat pump cycles that are frequently discussed in the literature. These heat pump systems can produce hot water at temperatures up to 95°C from wastewater with a mean temperature of 45°C. The results demonstrated that a two-stage heat pump with a flash vessel and a two-stage heat pump with a flash vessel as well as an intercooler had significantly less exergy destruction than an SS cycle. Both cycles showed a 10 % increase in overall exergy efficiency compared to the SS cycle. Similarly, Arpagaus et al. [14] compared the performance of several multi-temperature heat pump cycles based on designs mentioned in literature and concluded that multi-stage compressor cycles exhibit the best cycle performance as well. However these authors only evaluate the performance at a cycle-level and do not examine the origin of the increased efficiency on a component-level.

The performance of individual components was examined by researchers such as Bergamini et al. [12], Hu et al. [15] and Mateu-Royo et al. [16]. Bergamini et al. [12] used exergy analysis to study the exergy destruction in a high-temperature single-stage ammonia heat pump that produced heat up to 140°C from an isothermal 30°C source. They found that the exergy destruction in components increased at different rates. For instance, the exergy destruction in the expansion valves was the most sensitive to the temperature lift and was the highest of all components at high temperature lifts, whereas its contribution to the total exergy destruction was limited at low temperature lifts. Hence, the dominant source of exergy destruction in the heat pump varied among components as the temperature lift increased. Hu et al. [15] found similar results when assessing a multi-stage heat pump that produced pressurized water at 120°C from a heat source ranging from 50 to 90 °C. Mateu-Royo et al. [16] used exergy analysis to identify possible further improvements to an experimental IHX-cycle setup for heat sinks up to 140°C. They found that the expansion valve had the lowest exergy efficiency, whereas the compressor had the largest contribution to the total exergy destruction. Whether, design changes would improve the techno-economic performance of these heat pump cycles is not explored in these studies. Yet, exergy analysis provide a basis for economic assessment of a heat pump. For instance, Farshi et al. [17] used exergy-based cost analysis to compare a novel ejector-boosted hybrid heat pump with existing absorption, compression, and absorption-compression heat pumps. Their results show that the newly proposed design provides clear techno-economic advantages over the reference cycles at high temperature lifts. Wang et al. [18] applied the exergy-based economic analysis to compare the performance of mechanically and thermally driven heat pumps. They demonstrate that exergy loss per capital investment, as a function of temperature lift, differs between mechanically and thermally driven heat pumps. Based on this distinction, they formulated a guide map to aid technology selection. In a follow-up study, Wang et al. [19] used the same principles when evaluating the performance of a transcritical heat pump cycle for hot water production and increased the COP of an originally single-stage transcritical cycle by 7 % based on design suggestions made in the literature.

The aforementioned studies of high-temperature and/or steam-generating heat pumps either compare previously described heat pump cycles from the literature or use these literature-based cycles to benchmark new cycle layouts. However, they do not demonstrate how exergy-based methods can be used to identify techno-economic design improvements for high-temperature applications and to aid in identifying design improvements. Tsatsaronis and Moran [20] addressed this gap for energy-conversion technologies overall by proposing the method of exergy-aided cost minimization. In their method, they rank the components of a system in descending order of combined cost of investment and exergy destruction. Subsequently, they use this ranking to evaluate how much an additional component would increase both the technical and economic performances.

This study aims to demonstrate how exergy-based cost minimization can systematically identify techno-economic improvements for a steam-generating heat pump, adding a new perspective to the limited body of literature on this topic. The method was illustrated with a case study where 10 tonnes per hour of 2 bar(a) steam are produced from 50 kg/s of wastewater with an initial temperature of 80°C. This case study was chosen due to its relevance across various industries, including chemical, paper, and food production. All configurations were modelled using refrigerant R-1234ze(Z) due to its high critical point, low global-warming potential, and ozone depletion potential [21].

2. Method

This section outlines the systematic approach used to identify techno-economic improvements for steam-generating heat pumps. Section 2.1 provides a general description of the overall methodology, including the key steps involved. Section 2.2 details the thermodynamic analysis, explaining the principles and calculations used to assess the heat pump's performance. Section 2.3 focuses on the economic evaluation, describing the cost calculations and economic indicators considered.

2.1. *Identification of techno-economic improvements*

Advancements to a heat pump cycle were explored by using the following four-step method: 1. setting cost targets, 2. performing energy, economic and exergy analysis (3E-analysis), 3. assigning costs to exergy losses, and 4. assessment of design changes. Step 1 was used to make an initial assessment of a heat pump's economic viability compared to an electric boiler (e-boiler). When the resulting investment budget for a heat pump seemed plausible, the next step was initiated. Steps 2 to 4 were part of an iterative loop to repeatedly improve the cycle's performances. For clarity purposes, this loop was demonstrated only once for a SS cycle.

Step 1: To set cost targets, the economic performance of an ideal (i.e., Carnot) heat pump was compared with an e-boiler. This comparison is common in industry because both are a way to realise industrial electrification. The heat pump was defined to be economically viable when the total costs of ownership (TCO) after five years of full-time operation (8000 h) were lower than that of an e-boiler. The cost of electricity was taken to be 0.041 €/kWh based on the expected average electricity costs between 2022 and 2030 in the Netherlands for large consumers [22]. The required capital investment for an e-boiler of the required size was based on an installed capital cost price of 165 €/kW [22]. For the heat pump, an installation cost factor of 3 was used to convert bare unit costs to installed costs [23]. Based on the TCO of an e-boiler and the operational costs of an ideal heat pump, the maximal capital investment for a heat pump was calculated. If the calculated maximal capital costs price of the heat pump was within a realistic cost price range (100–1000 €/kW) [24], the next step was initiated.

Step 2: The energy, economic and exergy (3-E) analysis of the heat pump cycle was performed as follows. The thermodynamic states of the heat pump were fixed by the outlet conditions of both the evaporator and the condenser. For both heat exchangers the pinch point temperature difference was set to 5 K [25]. The vapour exiting the evaporator and the condenser was assumed to be saturated. When subcooling or superheating were considered, the amount of heat transferred was limited by the temperature at the outlet of the compressor ($<$ 175°C) to limit the degradation of compressor lubricants and seals [10]. An isentropic efficiency of 70 % was used for the compressor [26]. The compressor's drive operated with an overall motor efficiency of 85 % [25]. The costs of the bare units, e.g., the heat exchangers, were based on their duty and cost functions. The bare unit costs were indexed to December 2022 with the Chemical Engineering Price Index (CEPI) [27] and converted into a total capital investment (TCI) using an installation factor to account for the cost of integrating the unit, contingencies and other fees. The performance of the cycle was defined based on four performance indicators: 1. the total costs of ownership, 2. the initial investment, 3. the coefficient of performance, and 4. the total exergy destruction. Of these, the first two indicated economic viability, whereas the third and fourth gave insight into the technical and environmental performance of the proposed configuration, respectively.

Step 3: The exergy-based cost minimization was based the exergy-aided cost minimization algorithm by Tsatsaronis and Moran [20], where the exergy flow rate and is used to calculate the cost flow rate. Since the exergy destruction in the heat pump results in additional work requirements that have to be met by the compressor's drive, the price of exergy destruction was uniformly set to that of electricity. The resulting costs per component were listed in descending order. The process causing the losses in the top ranking component was selected to be changed in step 4.

Step 4: The design change was realized by adding one or several of the standard cycle's components, i.e., a compressor, an expansion valve, an internal heat exchanger, an ejector, a flash tank, a desuperheater, a cascade condenser, and/or an expander. The selection among these components was based on the origin of the exergy destruction and the estimated costs of the design change. Moreover, the way these components were integrated into the cycle was based on the cycles presented in the aforementioned studies by Adamson et al. [9], Arpagaus et al. [3,14], Mateu-Royo et al. [10], and Schlosser et al. [28].

After changing the cycle configuration in step 4, step 2 was repeated to evaluate the techno-economic performance of the change in the design. The change was approved when it improved the techno-economic performance of the previously evaluated heat pump configuration, which was followed by suggesting further improvements based on repeating steps 3 and 4. When the change of step 4 was disapproved in step 2, the design change was discarded and the iterative optimization was ended.

Cycle improvements were also based on an energy analysis to benchmark the results of the exergy-based cost minimization with a more common approach.

2.2. *Thermodynamic analysis*

2.2.1. *Energy analysis*

The basis of the energy balance was a consistent mass balance. The mass flow rate of the refrigerant (m_r) was defined by the heat transfer required in the condenser (Q_{cd}) and the enthalpy difference (Δh_{cd}) over the condenser, as shown in Eq. (1):

$$Q_{cd} = m_r \Delta h_{cd} \tag{1}$$

The refrigerant exited the condenser as a saturated liquid. All open systems were assumed to operate in a steady state, thus without mass accumulation. This also holds in the case of a (flash) vessel, where the vapor left in a saturated state and an enthalpy balance defined the mass ratio of its outgoing streams.

Work added to the system (W_c) by the compressor was based on the isentropic enthalpy difference (Δh_{is}) over the compressor and an isentropic efficiency (η_{is}) [26,29], as indicated in Eq. (2):

$$W_c = \frac{m_r \Delta h_{is}}{\eta_{is}} \tag{2}$$

The work required by the compressor's drive (W_D) was based on a correction for electrical, volumetric, and mechanical losses based on overall motor efficiency (η_m) [25], as shown in Eq. (3):

$$W_D = \frac{W_c}{\eta_m} \tag{3}$$

The intermediate pressure (p_i) was corrected by 0.35 bar when multiple pressure stages were considered based on the work by Mateu-Royo et al. [30], as in Eq. (4):

$$p_i = \sqrt{p_1 p_2} + 0.35 \quad (4)$$

where p_1 and p_2 are the pressures before and after the compressor, respectively. Pressure relief in expansion valves was considered isenthalpic. Other forms of pressure loss were neglected, as well as heat losses. The coefficient of performance (COP) of the heat pump was based on the heat delivered at the condenser and the work required by the compressor's drive, as shown in Eq. (5):

$$COP_{hp} = Q_{cd}/W_D \quad (5)$$

The energy balance of the heat pump was closed by defining the required thermal duty of the evaporator as the difference between the duties of the condenser and the compressor.

2.2.2. Exergy analysis

The influx of exergy ($\dot{Ex}_{in}$) equals the outflux of exergy ($\dot{Ex}_{out}$) plus exergy losses. The loss of exergy was the sum of internal exergy destruction (Ex_{des}) and transfer of exergy to external sources [31]. Since heat loss to the environment was neglected and all heat transferred from the heat pump to the environment was valuable, the exergy balance simplifies to Eq. (6):

$$\dot{Ex}_{des} = \Sigma \dot{Ex}_{in} - \Sigma \dot{Ex}_{out} \quad (6)$$

Exergy destruction was zero in the case of an ideal operation. In that case, the exergy flowing into the system in the form of heat at the evaporator and work by the compressor was equivalent to the exergy of the outflow of heat at the condenser. The exergy value of the streams was defined by the enthalpy (H) and entropy (S) of the stream shown in Eq. (7) [32]:

$$Ex = H - H_0 - T_0(S - S_0) \quad (7)$$

where subscript "0″ denoted the reference state at $T_0 = 298{,}15$ K and $p_0 = 101325$ Pa. Substituting Eq. (7) in Eq. (6) and accounting for the exergy value of heat: $Q(1 - T_0/T)$ at a thermodynamic mean temperature and that of work: W, results in Eq. (8):

$$\dot{Ex}_{des} = \left(1 - \frac{T_0}{T}\right)\dot{Q} + \dot{W} - \dot{m}[h_2 - h_1 - T_0(s_2 - s_1)] \quad (8)$$

Exergy destruction due to mechanical losses in the drive was taken as equivalent to the loss of work during transfer.

2.3. Economic evaluation

The total cost of ownership and the total capital investment were taken as the key performance indicators for the economic evaluations. These costs were based on the indexed bare unit costs of the components, the cost of installation, and operation. The bare unit costs ($C_{0,p}$) required for the heat pump's components were based on the costs function provided by Zühlsdorf et al. [25], which is presented in Eq (9):

$$\log(C_{0,p}) = k_1 + k_2 \log x + k_3 (\log x)^2 \quad (9)$$

where "x" is the scaling parameter of a certain technology and "k_i" is a calibrated value. Table 1 shows the used values adapted from Zühlsdorf et al. [25].

The values in this table were harmonized into the equivalent costs of the components for December 2022 with f_{cepi} based on the Chemical Engineering Price Index (CEPI) [33]. That is, the indexed bare unit cost ($C_{0,ind}$) were calculated according to Eq. (10):

$$C_{0,p,ind} = C_{0,p} f_{cepi} \quad (10)$$

The indexed bare module costs were converted into capital investment (CI) using the installation costs factor (f_{IF}), as shown in Eq. (11):

$$CI_p = C_{0,p,ind} f_{IF} \quad (11)$$

The total capital investment (TCI) was calculated by taking the sum of all components in the configuration. For benchmarking purposes, this value is expressed as a factor of the condenser duty. For the centrifugal compressor and its drive, Eq. (4) was used as input for the scaling parameter by either including or excluding η_m, respectively. The resulting costs were benchmarked to the cost data provided in the DACE booklet [34] and found to be plausible. In case multiple compressors were used, their scaling parameters were combined to account for the economics of scale. Their respective costs were based on the ratio between the scaling factor of the individual component and the scaling factor of the combined components. The volume of a vessel was based on being able to supply the outlet streams for 10 min without an influx of new refrigerant. The heat exchanging area (A) of the heat exchangers was calculated using Eq. (12):

$$Q_{hx} = U \cdot A \cdot \Delta T_{lm} \quad (12)$$

where "U" is the heat transfer coefficient and ΔT_{lm} was the logarithmic mean temperature difference between the hot and cold streams. A heat transfer coefficient of 1000 W/m^2K was used for heat transfer between a liquid and an evaporating liquid and 1250 W/m^2K was used when both sides were changing phases [25]. The operational cost ($C_{op,hp}$) of the heat pump (hp) was defined to be equivalent to the price of electricity (c_{el}) times the exergy destruction by component (p) and time of operation (t), as shown in Eq. (13):

$$C_{op,hp} = \sum \dot{Ex}_{des,p} \cdot c_{el} \cdot t \quad (13)$$

From Eq. (13), the cost of compensating irreversibilities (i) required due to exergy destruction of component "p" ($\dot{C_{i,p}}$) can be calculated as Eq. (14):

$$\dot{C_{i,p}} = \dot{Ex}_{des,p} \cdot c_{el} \cdot t \quad (14)$$

Economically viable investments were those with an investment cost lower than savings according to Eq. (14) after five years of full-time operation. Moreso, by combining the TCI and the $C_{op,hp}$ the total cost of ownership (TCO) was calculated, as in Eq. (15):

$$TCO = TCI + C_{op,hp} \quad (15)$$

The TCO was calculated based on five years of near full-time operation (40,000 h).

Table 1
Parameters for estimation of component capital costs according to Zühlsdorf et al. [25].

Component	Scaling Parameter X	Range	k_1	k_2	k_3	f_{cepi} [33]	f_{if}
Compressor	Fluid power	450–3000 kW	2.2897	1.13604	−0.1027	2.3749	2.8
Drive	Shaft power	75–2600 kW	1.9560	1.7142	−0.2282	2.3749	1.5
Plain vessel	Volume	1–800 m^3	3.5970	0.2163	0.0934	2.0793	3.0
Shell & tube heat exch.	Area	10–900 m^2	3.2476	0.2264	0.0953	2.0793	3.2
Radial turbine	Fluid power	100–1500 kW	2.2476	1.4965	−0.1618	2.3749	3.5

3. Results

This section presents the findings from the techno-economic analysis of steam-generating heat pumps. Section 3.1 compares the economic viability of an ideal heat pump with that of an electric boiler (e-boiler), establishing a baseline for comparison. Section 3.2 provides the results of the energy, economic, and exergy (3-E) analysis of the subcritical single-stage (SS) cycle, highlighting key performance metrics. Section 3.3 evaluates the performance of proposed cycle improvements, detailing the enhancements and their impacts. Finally, Section 3.4 compares the different configurations, summarizing the overall findings and identifying the most effective design modifications.

3.1. Cost targets for a steam-generating heat pump

Producing 10 t/h of 2.0 bar(a) steam with an e-boiler required 6.6 MW of electricity. Based on the assumed installed costs of 165 €/kW and an electricity price of 0.041 €/kWh, this resulted in a total cost of ownership (TCO) of 11.9 M€ after 5 years, as listed in Table 2. The total costs of ownership of the heat pump must be below that of the e-boiler to be competitive. An ideal heat pump would operate with a COP of 6.6 and reduce electricity consumption by 85 %. As a result, it would require 1.0 MW to operate, or 1.6 M€ after five years. Therefore, the total installed costs of the heat pump must be below 10.3 M€, or 520 €/kW$_{th}$, to be economically viable.

3.2. Energy, exergy, and economic performance assessment of a subcritical single-stage heat pump

The SS cycle consists of an evaporator, compressor, condenser and expansion valve. The results of the 3-E analysis of this cycle are presented in Table 3. These results show that the compressor's drive is the largest energy consumer. The electric drive of the compressor required 2.9 MW and the COP was therefore 2.3. Hence, compared to the ideal heat pump's 1.0 MW electricity consumption, 1.9 MW is required to compensate for exergy destruction. No costs were assigned to the expansion valve as the required capital investment was two orders of magnitude less than that of the other components. The compressor and its electric drive cost 3.2 M€, making up for more than 70 % of the total installed costs. Total costs of ownership (TCO) of 9.2 M€ were based on the electricity consumption of the drive and the total installed costs. The indexed bare unit cost of the module was 233 €/kW.

The results presented in Table 3 highlight that energy is solely required by the compressor and its drive. Consequently, enhancing the compressor's efficiency emerges as a logical solution based on the energy analysis. The table also shows that the expansion valve is responsible for most of the exergy destruction, accounting for 689 kW (or 37 % of total exergy destroyed) and incurring operational losses of over 1.1 M€ after 5 years of operation. The compressor is the second largest source of exergy destruction (23 % of total exergy destroyed at 545 kW) and adds 0.9 M€ to the operational losses after 5 years. The impact of the heat exchangers on the operational cost is below 5 %. Consequently, utilizing the exergetic potential of the stream before the expansion valve whilst trying to improve on the performance of the compressor should be pursued based on the results of the exergy analysis.

3.3. Improving the subcritical single-stage heat pump configuration

The results of the energy analysis indicated that increasing the efficiency of the compressor is pursued with a two-stage compression cycle with intermediate cooling (2IC), as depicted in Fig. 1.A.

The results of the exergy analysis indicated that utilizing the exergy before the expansion valve. Utilization of exergy in another process step can either be achieved by exergy transfer through heat exchange or mixing. The use of an internal heat exchanger (IHX) (Fig. 1.B) is a cost effective way to exchange heat [35]. Mixing is realized in the two-stage compression with a flash vessel (2FV) cycle (Fig. 1.C). This cycle also increases the compressor's efficiency by reducing the inlet temperature of the high pressure compression stage [16]. For this reason, this cycle design is selected over the additionally listed two-stage compression cycle with a closed economizer by Adamson et al. [9], as that option does not have this benefit. Another benefit of the 2FV cycle is that many industrial scale compressors allow for some form of two-stage compression within the same compressor unit and hence two-stage compression does not significantly increase equipment cost when applying intermediate cooling, something that would not be possible with a cascade cycle [6].

3.3.1. Subcritical two-stage compression with intermediate cooling cycle

The introduction of a second compression stage with an intermediate cooling to transform the SS cycle into the 2IC cycle, configuration A, reduces the COP from 2.3 to 2.2. Exergy losses are dominated by the expansion valve at 641 kW, or 32 % (Table 4). However, combined losses in both compressors and drives account for 1,042 kW of the total 2,016 kW exergy destroyed, i.e. 53 %. Though the specific work requirements by the compressor were slightly reduced by the intermediate cooling step, these gains are negated by the required increase in refrigerant mass flow to meet the energy demand in the condenser. This is partially a result of not being able to utilize the apparent heat in the intercooler due to its relatively low temperature of 89 – 92°C and an initial sink temperature of 80°C with an advised minimal temperature difference of at least 5 K [25]. Due to the higher work requirements and the additional investment, the total cost of ownership of this configuration is higher than that of the original SS cycle. The total installed costs of the configuration are 4.9 M€, with a TCO of 9.4 M€, or 254 €/kW as a bare module.

3.3.2. Subcritical single stage with internal heat exchanger cycle

The addition of an internal heat exchanger (IHX) to the SS cycle, to form the IHX cycle (configuration B), significantly increases the overall performance of the heat pump, leading to a reduction in the total costs of ownership (TCO) by 0.5 M€ compared to the SS cycle. Although the installation of the IHX increases the initial installed costs by 0.5 M€, the reduction in the size of the compressor and electric drive results in savings of 135 k€ in installed costs and a reduction of 0.8 M€ in operational costs after 5 years (Table 5). The total cost of ownership for the heat pump with the IHX is 8.7 M€ or 248 €/kW as a bare module.

The lower operational cost are reflected in the increase in the COP from 2.3 to 2.8. The total exergy destruction is reduced by 490 kW to 1,358 kW. The compressor is the main source of exergy destruction at 379 kW (28 % of the total exergy destruction), followed by its driver at 351 kW (25 %) and the condenser at 290 kW (22 %). The exergy destruction becomes more evenly distributed among the components

Table 2
Costs comparison of an ideal e-boiler with an ideal heat pump based on 5 years of operation, 8000 h/year, and an electricity price of 0.041 €/kWh.

Unit	Supplied power [MW$_{th}$]	Required power [MW$_e$]	Specific costs [€/kW$_x$]	Bare unit costs [M€]	Total installed costs [M€]	Operational costs [M€]	Total costs of ownership [M€]
e-boiler	6.6	6.6	–	–	1.1	10.8	11.9
Heat pump	6.6	1.0	<520	<3,4	<10.3	<1.6	<11.9

Table 3
3E-evaluation of a SS cycle heat pump based on 5 years of operation, 8000 h/year, and an electricity price of 0.041 €/kWh.

Component	Heat transfer [MW]	Required power [MW]	Exergy destruction [kW]	Operational losses [k€]	Scaling factor [X]	Indexed bare unit costs [k€]	TCI [k€]	TCO [M€]
Evaporator	4.1	0.0	95	156	278	161	514	
Compressor	0.0	2.4	545	894	2,448	724	2,028	
Drive	0.0	2.9	432	708	2,880	412	1,153	
Condenser	6.6	0.0	87	142	1,054	248	792	
Exp. Valve	0.0	0.0	689	1,131				
Total			1,848	3,031		1,544	4,488	9.2

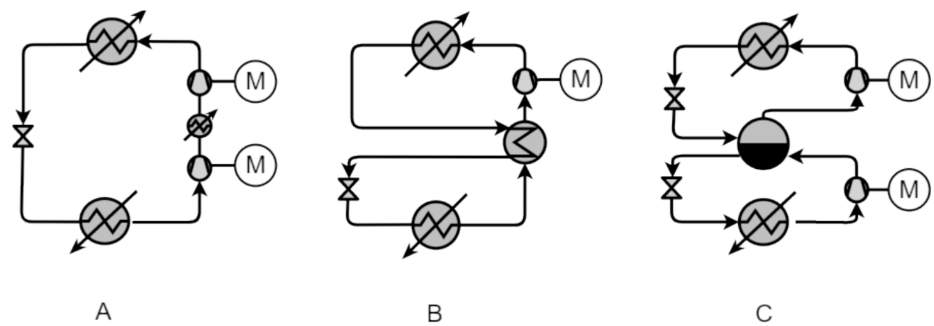

Fig. 1. Overview of advanced configurations. A) two-stage compression and intermediate cooler (2IC) cycle, B) Internal heat exchanger (IHX) cycle, and C) two-stage compression with a flash vessel (2FV) cycle.

Table 4
3E-evaluation of the two-stage compression and intermediate cooling cycle based on 5 years of operation, 8000 h/year, and an electricity price of 0.041 €/kWh.

Component	Heat transfer [MW]	Required power [MW]	Exergy destruction [kW]	Operational losses [k€]	Scaling factor [X]	Indexed bare unit costs [k€]	TCI [k€]	TCO [M€]
Evaporator	4.3	0.0	174	286	282	161	516	
Compressor 1	0.0	1.4	343	563	1,392	397	1,112	
Drive 1	0.0	1.6	246	403	1,637	223	625	
intercooler	0.3	0.0	55	90	102	121	388	
Compressor 2	0.0	1.2	264	433	1,183	338	946	
Drive 2	0.0	1.4	209	342	1,392	190	530	
Condenser	6.6	0.0	84	138	1,054	248	792	
Exp. Valve	0.0	0.0	641	1,052				
Total			2,016	3,307		1,678	4,909	9.4

Table 5
3E-evaluation of the internal heat exchanger cycle based on 5 years of operation, 8000 h/year, and an electricity price of 0.041 €/kWh.

Component	Heat transfer [MW]	Required power [MW]	Exergy destruction [kW]	Operational losses [k€]	Scaling factor [X]	Indexed bare unit costs [k€]	TCI [k€]	TCO [M€]
Evaporator	4.6	0.0	195	319	288	162	520	
Compressor	0.0	2.0	379	621	1,991	682	1,909	
Drive	0.0	2.3	351	576	2,342	406	1,137	
IHX	2.3	0.0	64	105	185	143	456	
Condenser	6.6	0.0	290	476	1,054	248	792	
Exp. Valve	0.0	0.0	79	130				
Total			1,358	2,227		1,640	4,814	8.7

with the introduction of the IHX. The increase in exergy destruction in the evaporator is due to the higher COP, requires more energy from the sink and results in a temperature drop from 55 to 54°C of the source. The exergy destruction in the condenser significantly increases due to its high inlet temperature, which is caused by superheating the suction gas before the compressor with the IHX. The IHX itself has an exergy destruction of 64 kW.

3.3.3. *Subcritical two-stage compression with a flash vessel cycle*

The two-stage compression with flash vessel (2FV) cycle, configuration C, splits work requirements over two compressors with a combined duty of 2 MW, or 2.4 MW at the electric drive (Table 6). Hence, the required 6.6 MW at the condenser can be delivered with a COP of 2.8. Exergy destruction is evenly distributed among the components. The second stage compressor was the main source of exergy destruction at 269 kW (19 % of total exergy destroyed), followed by its drive at 212 kW (15 %) and the expansion valve directly after the condenser at 207 kW (15 %). Together with the first stage and their drives, the compressors accounted for 60 % of total exergy destruction, compared to 21 % of both expansion valves. The exergy destruction in the evaporator increased by 100 kW as more heat was transferred. The intermediate cooling in the vessel slightly reduced exergy destruction in

Table 6
3E-evaluation of the two-stage compression and a flash vessel cycle based on 5 years of operation, 8000 h/year, and an electricity price of 0.041 €/kWh.

Component	Heat transfer [MW]	Work transfer [MW]	Exergy destruction [kW]	Operational losses [k€]	Scaling factor [X]	Indexed bare unit costs [k€]	TCI [k€]	TCO [M€]
Evaporator	4.5	0	192	314	287	162	519	
Compressor 1.	0	0.8	204	335	826	279	781	
Driver 1.	0	1.0	146	239	972	165	463	
Vessel	5.4	0	1	2	690	183	548	
Compressor 2.	0	1.2	269	442	1,207	407	1,140	
Driver 2.	0	1.4	213	349	1,420	241	676	
Condenser	6.6	0	84	138	1,054	248	792	
Exp. Valve 1.	0	0	207	339				
Exp. Valve 2.	0	0	87.4	143				
Total			1,403	2,301		1,685	4,919	8.8

the condenser compared to the SS cycle. The vessel itself has a negligible amount of exergy destruction. Exergy destruction in the expansion valves went from 689 kW in the SS cycle to 294 kW for both valves. As a result, total exergy destruction was reduced by 445 kW. The reduced size of the compressors and electric drive reduced investment costs by 43 k€. However, the installation of the vessel requires an additional 0.55 M€. The TCO is 8.8 M€, or 255 €/kW as a bare module. Hence, the initial investment increases compared to the SS cycle, but the increased efficiency mitigates the impact of operational costs and reduces the TCO by 0.4 M€ during the five years.

3.4. Comparison of configurations

The results presented in Tables 4–7 are visualized in Fig. 2. The left figure shows that all advanced cycles have a lower total exergy destruction than the SS cycle. The figure also shows the exergy destruction of the different components within the heat pump and, for example, the strong increase in exergy destruction at the evaporators of the advanced cycles due to the increased thermal demand. The figure also shows that the two exergy-based designs significantly reduced the exergy destruction in the expansion valve compared to the SS and the 2IC cycle. The IHX performed by far the best in this aspect. However, this is largely offset by the increase in exergy destruction in the condenser due to the high outlet temperature of the compressor. Due to this offset, the total exergy destruction by the IHX and the 2FV cycle are comparable. The exergy destruction in the 2FV cycle is more distributed among its components, which provides a lower basis for the next design iteration. Thermodynamically, the IHX cycle performs similar to the 2FV cycle, as the 2FV cycle has only a slightly higher total exergy destruction of 1,403 kW compared to 1,358 kW of the IHX cycle.

The figure on the right shows the required total capital investment of the four cycles on the primary y-axis and on the secondary y-axis their total cost of ownership. The graph shows the dominance of the compressor(s) and its/their drives in the total capital investment. The figure also shows that the IHX cycle economically outperforms the 2IC cycle based on by the energy analysis. The TCI of the IHX cycle is comparable to that of the 2FV cycle, as the 2FV cycle has a slightly higher TCI of 4,919 k€ versus 4,814 k€ of the IHX cycle.

Combined, the two graphs in Fig. 2 show that the designs based on exergy-based cost minimization have a higher techno-economic performance than the designs based on energy analysis. Moreover, the design change in the 2IC cycle, based on by the energy analysis, increased exergy destruction whilst increasing the TCO to 9.4. The design changes in the exergy-based cost minimization, on the other hand, had a lower TCO than the SS cycle. Both design changes increased the SS cycle's COP from 2.3 to 2.8. The introduction of the IHX increased the total capital invested by 330 k€ whilst reducing the total cost of ownership after 5 years by 0.5 M€. The 2FV cycle was slightly more costly and required an additional 0.6 M€ investment, decreasing the reduction of the total cost of ownership to 0.4 M€ with respect to the SS

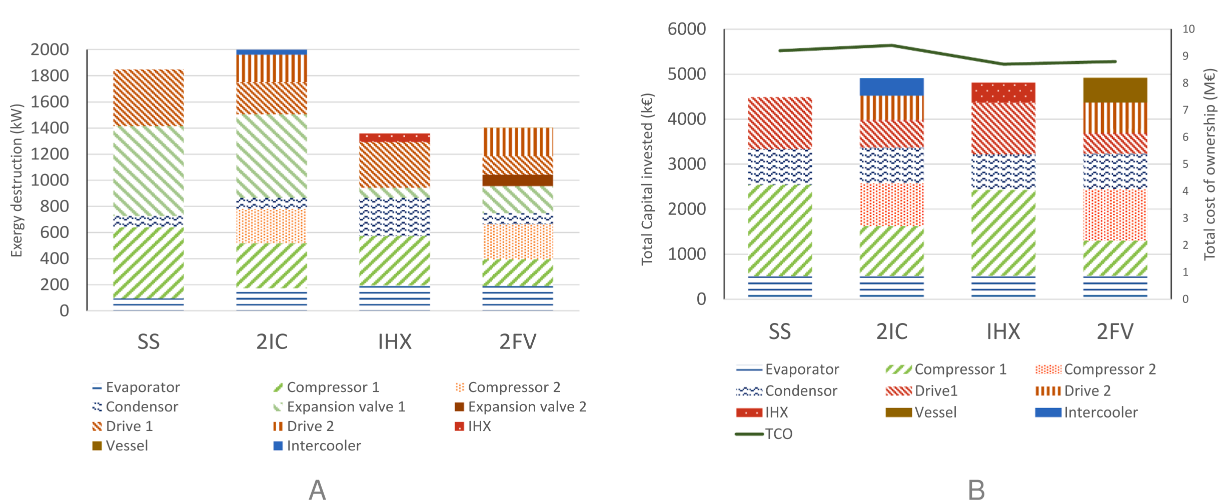

Fig. 2. Comparison of configurations. Figure A shows the exergy destruction by the subcritical single-stage (SS) cycle, compared to that of the two-stage cycle with an intermediate cooler (2IC), the cycle with an internal heat exchanger (IHX), and that of the heat pump with the two-stage compressor and flash vessel (2FV). Figure B shows the total capital invested and total cost of operation of these cycles.

cycle.

4. Discussion

The findings of this study are in line with the results of Bergamini [12] and Hu [21]. Similar to their work, our results showed that the expansion valve is the largest single source of exergy destruction in the SS cycle, followed by the compressor in steam-generating applications. Moreover, the COP of 2.3 for the SS cycle is comparable to the COPs based on experiments and simulations, i.e. 1.7 to 2.3, when transferring heat of $60 - 100^{\circ}$C to a heat sink of 140°C listed by Adamson et al. [9]. Moreso, the COP of the IHX cycle is likely of the right order, as it is slightly below the 3.05 reported by Wang et al. [36] based on experiments with a SGHP with a 70°C heat source. However, this improvement contradicts the listed performance by Adamson et al. [9] which shows that the IHX cycle does not significantly improve upon the SS's performance, which highlights the added value of the method proposed in this paper. Moreover, the COP of the heat pump with 2FV cycle is comparable the COP of 2.9 reported by Kosmadakis et al. [35] for the same process conditions using R1234ze(Z) and a numerical model for the efficiency of the compressor.

The results also show that the exergy-based improvements of the cycles outperform energy-based improvements due to their ability to utilize waste streams. The lower thermodynamic performance and high required investment cost of the 2CI cycle make it unlikely that changes in either operational or investment cost will alter this outcome. The techno-economic performance of the IHX cycle was comparable to that of the 2FV cycle, as there is no significant difference in the amount of total exergy destruction (1,403 kW for the 2FV cycle compared to 1,358 kW for the IHX cycle) and the total capital invested (TCI) (4,919 k€ for the 2FV cycle versus 4,814 k€ for the IHX cycle). Changes in modelling assumptions, assumed cost prices or installation cost factors might bring these performances even closer together or tip the scales. Herein, a key modelling assumption is working with a constant and relatively low isentropic efficiency for the compressor. The use of temperature dependent isentropic efficiencies, as proposed by Mateu-Royo et al. [10], might be a disadvantage for the IHX cycle where suction gas is preheated. However, this disadvantage might again be compensated by using a compressor with a higher isentropic efficiency, e.g. 80 % compared to the used 70 % [12]. The cost functions and installation cost factors are another key factor in this evaluation as the total indexed bare unit cost only differs by 40 k€ or 2 %.

Nonetheless, both solutions demonstrate that using insights from exergy-based cost minimization improve the techno-economic performance of SGHPs more than energy-based solutions. However, the method is limited because important decision aspects such as process dynamics, maintainability and environmental impact are not included. Another point to consider is that, although the method allows for the addition of top/bottom cycles, an extension of the method to identify when adding or changing the working media improves performance would be beneficial.

5. Conclusions and recommendations

An iterative method was developed to identify techno-economic improvements for a steam-generating heat pump. It was demonstrated that exergy-based cost minimization can be used as a systematic assessment tool to identify techno-economic improvements for steam-generating heat pumps (SGHPs). By applying this approach, the study adds a new perspective to the limited body of literature in exergy-based cost analysis of SGHPs. The method identified the most cost-effective design change to increase the techno-economic performance of the SS cycle. This is realized by adding an internal heat exchanger (IHX) and utilizing the exergetic potential of the stream after the condenser. This change increases the COP from 2.3 to 2.8 and lowers the total cost of ownership (TCO) from 9.2 M€ to 8.7 M€ after 5 years.

Furthermore, the results showed that exergy-based design changes resulted in a higher COP and lower TCO than energy-based ones. The results of the energy analysis indicated that improving the compressor efficiency with the help of intermediate cooling. The COP of this cycle decreased by 0.1, whilst increasing the TCO. The exergy-based design changes increased the SS cycle's COP from 2.3 to 2.8 at similar investments. Hence, the exergy-based cost minimization proved to be a better performing assessment tool than energy analysis to systematically identify improvement to SGHP cycles.

Future work should develop a more detailed thermodynamic analysis (e.g., using temperature dependent isentropic efficiencies) to improve the assessment of strengths and weaknesses of a cycle, provide a framework to link working media to exergy destruction, and expand on the considered aspects during the decision process (e.g., environmental footprint and process dynamics).

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Data availability

No data was used for the research described in the article.

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